Engineering Report No. 6

on

EXPERIMENTAL STUDY EVALUATING STRUCTURE-BORNE AND AIR-BORNE VIBRATIONS

Research

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EXPERIMENTAL STUDY EVALUATING STRUCTURE-BORNE AND AIR-BORNE VIBRATIONS

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This experimental study was performed under the direction of the undersigned with the assistance of Messrs. O. E. Curth, D. F. Muster, and L. Tybur. Other members of the staff took part from time to time as the need arose.

Respectfully submitted,

Chester A. Arentz

Professor Chester A. Arentz
Project Director

Chicago 16, Illinois
15 December 1953

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ABSTRACT

The purpose of this investigation was to determine the contribution of the air-coupled vibrational energy to the overall energy emanating from an enclosed vibrating source. A vibration source suspended in a steel tank was used to simulate a noisy machine mounted within an enclosed chamber.

The vibration exciter or driver was suspended in the tank in three ways: by a solid steel connection, by a resilient mounting, and by a wire.

Recordings were made of the sound intensity outside the tank as the vibration exciter was driven over the frequency range of 60 to 7000 cycles per second (cps). To eliminate the air-coupled energy the tank was evacuated and recordings made.

It was found that: (1) The air-borne vibrations were too low in intensity to affect the decibel (db) readings caused by the structure-borne vibrations when the exciter was mounted solidly to the steel tank. (2) The resilient mounting greatly attenuated the structure-borne vibrations. Using the resilient connection, the air-borne vibrations had little effect on the structure-borne vibrations over a frequency range up to 2000 cps, but from 2000 to 7000 cps, the air-borne vibrations were predominant and raised the decibel level. This shows that air-borne vibrations are significant in the high frequency range. (3) Using the resilient mounting or the wire suspension connections the resilient mounting did not attenuate structure-borne

vibrations above 2000 cps as one would normally expect using basic vibration theory. (4) At frequencies below 150 cps the ambient noise masks signals from the test tank so no conclusion can be made in this frequency range.

Although the apparatus for these tests was not designed to simulate a submerged submarine, important observations were made which bear further study and evaluation: (a) A solidly connected vibrating source produced the most noise outside the test tank and the air-borne vibrations within the tank did not contribute to the noise radiating from it. (b) A resilient mounting was effective in decreasing the noise level radiating from the test tank over the frequency range 60 to 7000 cps; however, at high frequencies above 2000 cps the air-borne vibrations tend to by-pass the resilient mounting and raise the noise level radiating from the tank. (c) The resilient mounting appeared to lose its isolation effectiveness in the high frequency range.

Although this study is being continued by Illinois Institute of Technology, the findings outlined herein should be correlated with results from other laboratories. This may lead to important contributions in naval design for a quiet operating submarine.

I. INTRODUCTION

Background

Recent resilient mounting research has resulted in extending the theory of vibration and noise isolation. Investigators have detected wave effects in mountings and established methods to predict their characteristics [1, 2, 3][1]. The effects of a nonrigid or "live" foundation have been investigated theoretically and experimentally [4, 5] to determine the effect of foundation resilience on the transmission characteristics or isolation effectiveness of a resilient mounted vibrating system.

During a study of the vibration and sound transmission characteristics of resilient mountings an interesting phenomena was observed. At certain frequencies the apparent sound transmission through the mounting increased tremendously. Further investigation showed the transmission was not through the mounting but through the air surrounding the mounting. This observation led to the present work and this report.

Scope

It is felt that air-borne transmission of sound around a mounting instead of through it might occur in actual field installations nullifying the effectiveness of resilient mountings in certain applications. This effect is of particular interest to the Navy in their program of noise reduction as a means of reducing the possibility of enemy

[1] Numbers in brackets refer to the bibliography.

detection. The experimental methods used to determine the contribution of this air-borne energy to the total energy emanated from a vibration source are described in this report.

II. DISCUSSION OF PROBLEM

Importance of Air-Borne Vibrations

Inqueries and a search of the literature revealed a wide difference of opinion as to the importance of the contribution of air-borne vibrations within a ship or submarine hull to the overall noise emanating from such a hull. On one hand the acoustics people claimed that the impedance mismatch occurring between the air chamber within a submarine and the water surrounding it is so high that it may be neglected. On the other hand, reports from submarine men indicated that under some conditions voices have been heard from one submerged vessel to another.

Sound Attenuation

The ratio of reflected to incident sound intensity at the boundary between two different media is given by

$$\frac{I_r}{I_i} = \left(\frac{\rho_1 c_1 - \rho_2 c_2}{\rho_1 c_1 + \rho_2 c_2}\right)^2$$

where I_r is the reflected intensity

I_i is the incident intensity

ρ is the density of the medium

c is the velocity of sound in the medium

and the subscripts refer to the media.

If we consider steel and water as the two media, a ratio of 0.86 is obtained, 14 per cent being transmitted. This is equivalent to a decibel loss of about 18.7 db.

Considering steel and air as the media gives a ratio of 0.99996 or a transmission of 0.004 per cent, a loss of 88 db.

These results would lead one to agree with the acoustic people, since a single steel panel between a noise source and water or air would have an attenuation in excess of 100 db. However, depending upon the damping of the panel, a resonant condition of the panel will reduce the attenuation. A submarine is a complex structure which has many air chambers and structural members capable of supporting a resonant condition.

<u>Importance of this Study</u>

The experimental results reported here are not intended to be taken as representative of the spectra obtained from any piece of equipment in a submarine, but merely as those obtained from the particular apparatus used. The purpose of these tests is to indicate the amount of air-coupled energy that can by-pass a resilient mounting and appear outside the containing structure.

III. EXPERIMENTAL APPARATUS AND PROCEDURE

General Description of Apparatus

An apparatus was assembled to evaluate this problem by determining the noise level outside a test tank in which a noise and vibration source is mounted. Rather than use a machine with its limited frequency spectrum as the energy source, an electromagnetic vibration exciter was used for excitation at any desired frequency. The method of eliminating air-borne vibrations through the use of a vacuum lends itself to this problem. The sound and vibration exciter was suspended, in various manners, in the tank and records taken of the sound pressure outside the tank as the exciter was driven through the audio frequency range both with and without air in the tank. The exciter was mounted to the diaphragm top or the tank in two ways: first, solidly connected by means of a steel shaft, and second, by a resilient mounting. In addition it was suspended in the tank on a wire which was supported external to the tank. The wire was isolated mechanically and electrically from the tank and exciter. The position of the exciter within the tank was maintained constant for all methods of suspension. The tank and methods of suspension are shown in Fig. 1 and Fig. 2.

Detailed Description of Test Setup

The apparatus used in shown schematically in Fig. 3. The microphone, tank, accelerometer and its cathode follower are shown within the anechoic room at the upper

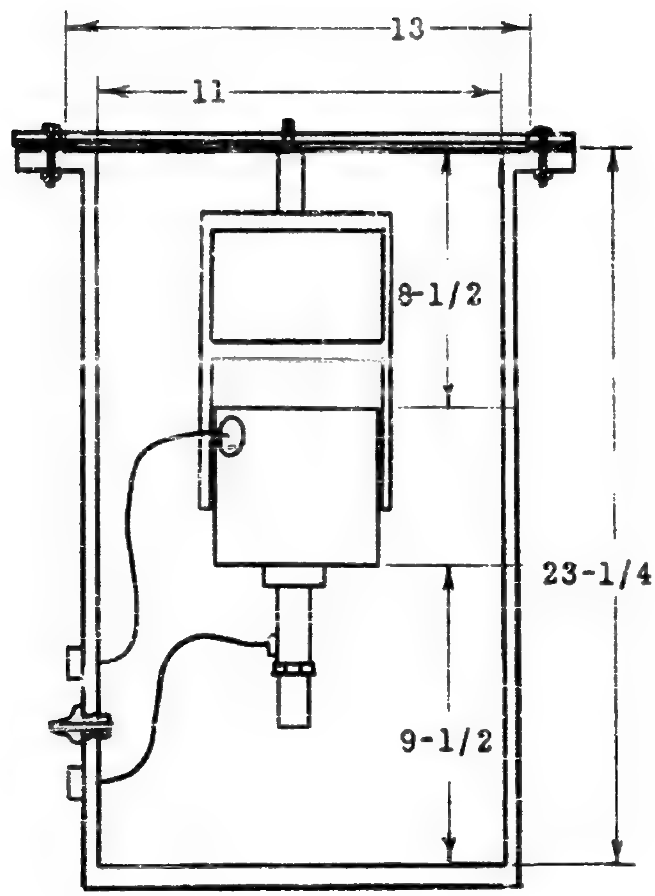


Fig. 1 Section Through Steel Tank

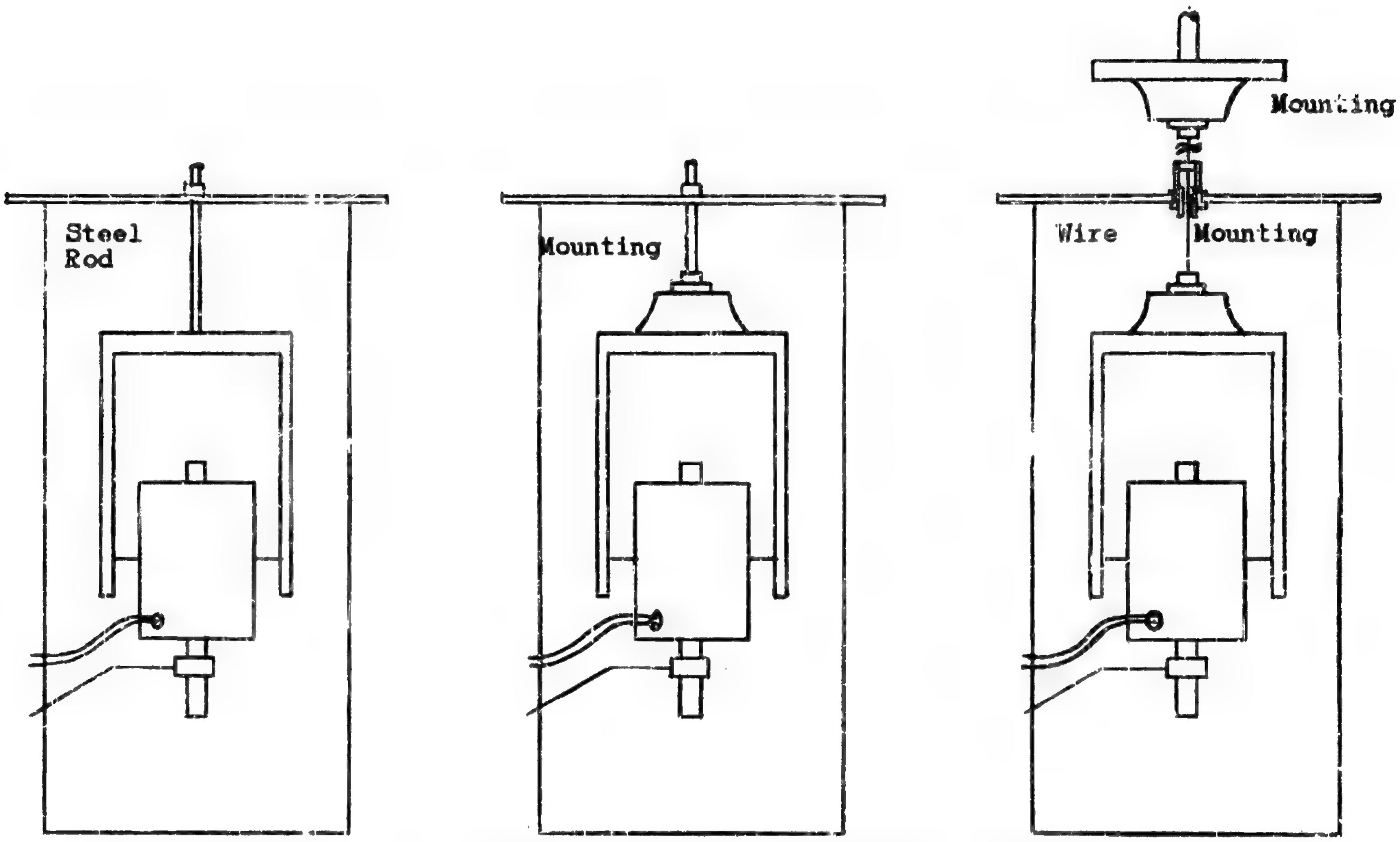


Fig. 2 Three Methods of Connecting Exciter to Diaphragm

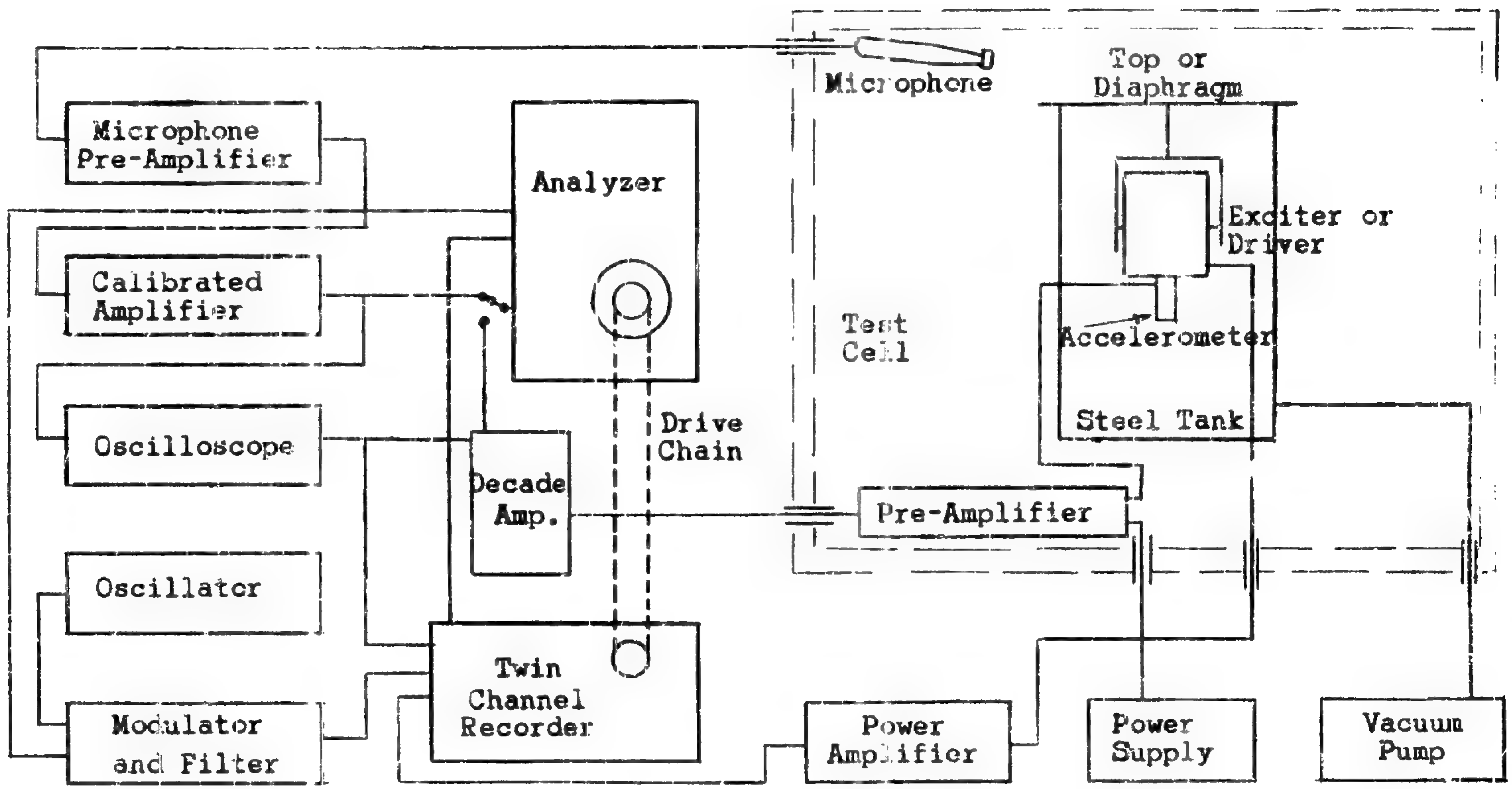


Fig. 3 Block Diagram of Test Apparatus

right, the rest of the instruments being external to the room for ease of operation.

The microphone, external to the tank, is used to detect the sound energy emanating from the tank. This sound was made up of the structure-borne and air-borne energy within the tank. Evacuating the tank attenuates the air-borne contribution and allows its effect to be noted. The tank is approximately two feet high and one foot in diameter with 3/8-inch walls and a 1/2-inch thick bottom. The tank top or diaphragm which supports the exciter is 1/8-inch thick. Provisions were made to support the exciter within the tank in three ways: (1) solidly connected to the diaphragm, (2) connected to the diaphragm through a resilient mounting, acting as the load for the mounting, and (3) isolated electrically and mechanically from the tank and diaphragm by a wire.

The solid connection allows readings to be taken which indicate the greatest energy which will leave the tank during these tests. Upon evacuating the tank the readings will indicate the structure-borne energy only.

The resilient mounting connection allows evaluation of the amount of energy by-passing the mounting, the difference between the readings observed with and without air in the tank being the object of these tests.

The third connection, with the exciter isolated from the tank, eliminated all structure-borne paths and allowed readings of the air-borne energy only. Readings with the

tank evacuated show the degree of success in mechanically and electrically isolating the system within the tank.

A Massa Model 117 ADP crystal accelerometer was mounted to the exciter to monitor the motion of the exciter.

Experimental Techniques

An early experimental technique consisted of driving the exciter through the desired frequency range simultaneously recording the wide-band output of the microphone and accelerometer with the twin channel recorder. As might be expected, both signals varied considerably, the forty decibel range of the recorder was exceeded by both signals. These variations were attributed to the variation in impedance with frequency of the mechanical items in the system. At some frequencies the sound output of the tank was very low, so low it was in the noise level of the room, while at other frequencies, resonant frequencies of the air chamber or of some mechanical part, the recorded amplitude would exceed the range of the recorder. These variations made comparisons of the recordings of the various setups almost impossible. Therefore, it was decided to use some sort of control circuit to maintain a constant signal output from the accelerometer. Several electronic methods were tried and discarded. Finally a method incorporating the recorder servo system, as used by Sykes and Harrison [3] was attempted and found satisfactory. Typical acceleration response curves are shown in Fig. 12, page 33. Since control had to be maintained at a level equal to or lower

than the lowest acceleration possible to obtain from the exciter in the frequency range used, it became necessary to use a narrow band analyzer to detect the desired signal and reject the unwanted noise of the test cell.

A Hewlett-Packard Model 300A harmonic analyzer was obtained for this use. To mechanically synchronize an oscillator with the Hewlett-Packard analyzer over the desired frequency range was almost impossible. Instead the analyzer oscillator was tapped and the voltage brought out to an external modulator where it was mixed with a fixed frequency voltage to obtain the desired driving frequency voltage for the power amplifier. A Hewlett-Packard 202B oscillator operated at 20 kc was used as the fixed oscillator. Undesired modulation products were attenuated by means of a filter.

The output of the filter was fed to the control circuit in the Sound Apparatus TFR recorder which operates as follows: The amplified accelerometer signal was fed to the amplifier and servo system of the recorder as shown in Fig. 4. The circuit was arranged so the servo system would move the potentiometer contact in a direction to maintain a constant signal at the accelerometer output. Oscillation of the entire network was damped by the TFR amplifier. The controlled signal was fed to the power amplifier which supplied energy to the exciter. The power amplifier was a "Williamson" type, employing a negative feedback loop from the output to the input stage resulting in a low output impedance.

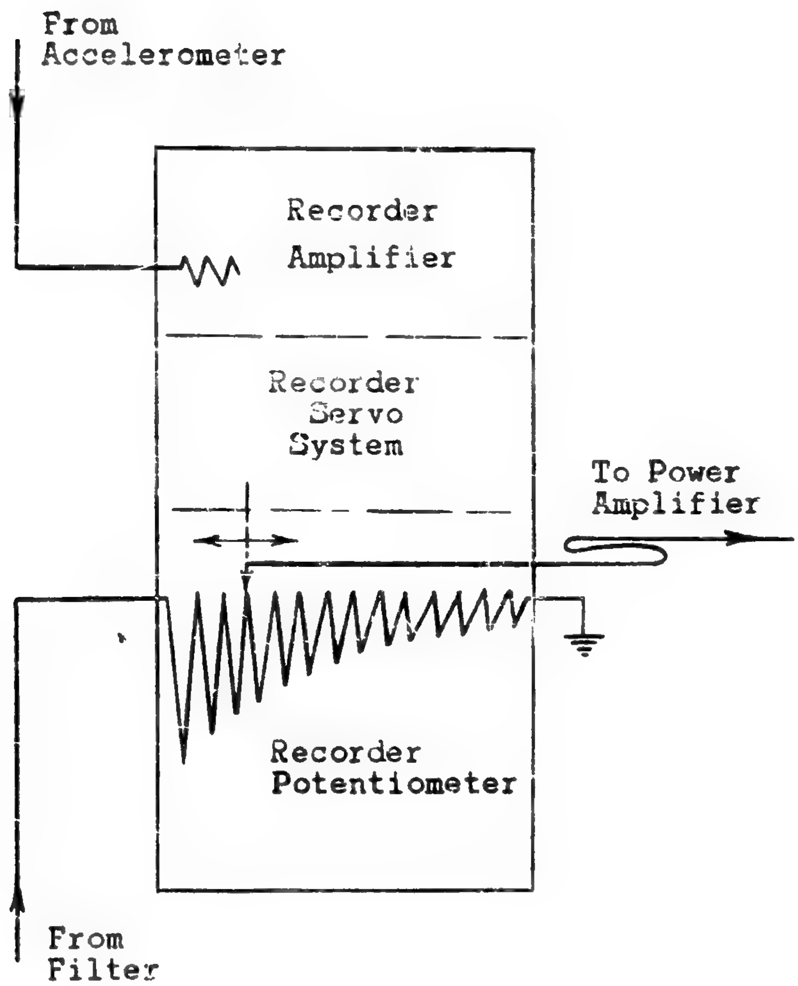


Fig. 4 Acceleration Control Circuit

Two transducers were used in the system, the previously mentioned Massa accelerometer and an Altec Model 21-B condenser type microphone. The output of the accelerometer was fed from its cathode follower pre-amplifier to a Ballantine decade amplifier and thence to the control system and also to the analyzer when desired. The microphone output was fed from its pre-amplifier to the calibrated amplifier. This amplifier supplies, depending upon the positions of its range switches, a signal of one volt (rms) for sound levels of 60, 80, 100 or 120 decibels (referred to 0.0002 dynes per square centimeter) at the microphone. The calibrated amplifier output was fed directly to the Hewlett-Packard analyzer.

Procedure in Recording Data

Microphone position in the test room was not found to be critical. During these tests the microphone position was maintained constant, two feet above and one foot off center of the tank diaphragm.

The steps used in recording the data were as follows:

1. Set up appropriate tank conditions, constant bolt torque.
2. Connect cables and allow electronic instruments to warm up.
3. Check gain, align frequency scale and balance the analyzer.
4. "Zero beat" fixed oscillator with the analyzer oscillator.
5. Start automatic recording.

Several recordings were made for each set of tank

conditions: (1) analyzed microphone signal (2) wideband microphone signal (3) analyzed accelerometer signal, and (4) wideband accelerometer signal.

Two sets of recordings were made, first with air in the tank and second with the tank evacuated. In most cases two sets of recordings were made in order to cover the full dynamic range required since the 40 db dynamic range of the recorder was not great enough. In all tests care was taken so as not to overload the instruments.

IV. TEST RESULTS

Sound pressure readings covering the frequency range from 60 to 150 cps are not considered accurate due to the wideband width used which passed low frequency signals from the building structure and any unbalance of the modulator in the analyzer, see Fig. 11. Therefore, in looking at the curves this frequency range should be disregarded.

<u>Exciter Solidly Connected to Diaphragm</u>

The purpose of solidly connecting the exciter directly to the diaphragm of the tank was to insure maximum transmission of structure-borne vibrations, see Fig. 2. Fig. 6 shows the response with air in the tank and with the tank evacuated. The solid curve represents readings with air, and the dotted curve represents readings in a vacuum. It can be seen that both curves are very much alike with one superimposed upon the other. The curves show that similar frequency components are present for both the air-filled and the evacuated tank. With the tank evacuated several discrete frequencies are shifted slightly to the right of those for air. This may be attributed to the change in diaphragm stiffness when the tank is evacuated and to the damping introduced by air in the tank.

Fig. 6 clearly indicates that the air-borne vibrations did not materially raise the decibel level of the sound emanating from the tank. However, this does not mean that the air-borne vibrations were not present. Referring to the test tank, if a discrete frequency peak shows a 52 db

reading in air as well as in vacuum, it does not mean that an air-borne vibration does not exist; in fact, an air-borne vibration could be present having a level of 42 db and still not influence the reading of a 52 db structure-borne vibration.

<u>Exciter Connected to Diaphragm Through a Resilient Mounting</u>

The purpose of connecting the exciter to the diaphragm through a resilient mounting was to check the effectiveness of a mounting in reducing the noise level and to see whether the air-borne vibrations within the tank were effective in raising the noise level caused by the structure-borne vibrations. Fig. 7 shows the response when the exciter is isolated from the tank diaphragm by means of a resilient mounting, Lord 204PH35, see Fig. 2. The solid line is for air and the dotted line is for vacuum. The transmissibility for this mounting is shown in Fig. 10. Referring to Fig. 7 it can be seen that the frequency peaks have shifted slightly to the left of those in air. Again this may be attributed to the change in diaphragm stiffness when the tank is evacuated and to the damping introduced by air in the tank. It can be seen that in most instances from 150 to 3000 cps the readings with air were about the same as those with a vacuum. However, in the frequency range above 3000 cps the readings with an air-filled tank were greater than those with a vacuum, showing that the air-borne vibrations within the tank supplemented the structure-borne vibrations to give a reading greater than the structure-borne vibrations alone.

Two questions naturally arise: (1) Why did the air-borne vibrations not add to the structure-borne vibrations to give greater readings when using the rigid connection (Fig. 6)? (2) Why did the air-borne vibrations not supplement those in air for the resilient suspension over the frequency range of 150 to 3000 cps (Fig. 7)?

The answer to the first question was given on page 15 and may be summed up as follows: When two sounds exist simultaneously and one is at least 10 db below the other, the lower sound pressure will not add to the higher to give an increase in the reading. Apparently the sound pressure level in air was about 10 db below or lower than the structure-borne vibrations thus indicating very little difference between the readings in air compared to those in vacuum.

The second question may be answered as follows: At low frequencies the wave lengths are large. Since the component parts of the driver are small they are not efficient in radiating the low frequency sound. As the frequency increases the wave lengths become smaller and the component parts become more efficient radiators of sound. The air-borne sound pressure level (decibel) rises and becomes great enough to raise the sound pressure level outside the tank above that caused by the structure-borne vibrations alone. Thus, when the air-borne vibrations are eliminated by the vacuum, the decibel levels drop below those with air. Therefore, the high frequency readings

above 3000 cps show that sound energy is by-passing the resilient mounting to give greater values with air than with a vacuum.

It is interesting to note that with the resilient mounting the attenuation of frequencies above 800 cps decreased. This can be explained by looking at the transmissibility curve, Fig. 10. It will be noted that at approximately eight hundred cycles per second this curve deviates from that expected by basic theory. At this point the isolator begins to lose its efficiency. Resonant peaks in the high frequency range are not attenuated to the extent one would normally expect. More will be said about discrete frequencies and transmissibility later on in this discussion.

Fig. 9 shows readings taken with air in the tank both for the rigid suspension and the resilient suspension. The solid line is for a rigid connection in air and the dashed line is for a resilient connection in air. The attenuation obtained by use of a resilient mounting is clearly shown. The attenuation varies but is as much as 40 db for some frequencies. The decrease in attenuation at the higher frequencies can be attributed to several factors: air-coupled energy by-passing the resilient mount, amplification due to resonances in the tank parts, and loss of isolation in the resilient mounting due to wave phenomena.

Exciter Suspended in Tank with Wire and Resilient Mountings

The driver was suspended in the test tank by means of

resilient mountings and a single strand of steel wire, see Fig. 2.

The purpose of this type of suspension was to eliminate, as far as possible, all structure-borne vibrations from the tank. Theoretically, the only sound waves emanating from the tank should be air-borne when the tank is filled with air. When the tank is evacuated, the microphone should indicate only the noise level in the room.

An examination of Fig. 8 shows that the sound pressure level was higher with air in the tank than with a vacuum. Due to an error in procedure no air-borne vibration data were taken below the 20 db level. As mentioned before, the moving parts of the exciter were small. Therefore, their effectiveness in coupling energy to air was slight at low frequencies and became greater in the frequency range above 2000 cps.

Fig. 8 shows that the vacuum was effective in eliminating the air-borne vibrations. At low frequencies the dotted curve representing the evacuated condition follows the noise level of the test cell, see Fig. 10. Although the noise level is below 20 db in the frequency range above 2000 cps, there is an increase in the sound pressure level. This may be explained by looking at the transmissibility curve for the resilient mountings used, see Fig. 10. The transmissibility curve shows that these mounts will pass certain high frequencies. Thus, one is lead to the conclusion that many of the high frequencies were passed with

enough energy to vibrate the supporting wire and other structural members to give a sound level rise above 2000 cps. This is true for both air and vacuum.

V. FREQUENCY ANALYSIS

It must be emphasized that it was not the purpose of this study to identify the discrete frequencies found with the use of the analyzer, but instead, to observe the over-all frequency spectra and note the effect of air-borne and structure-borne vibrations with and without the use of a resilient mounting.

A frequency analysis of the data becomes very difficult. The test tank with the driver mounted to a 1/8 inch steel diaphragm forms a complex vibrating system with many modes of vibration, see Fig. 1. Introduction of a resilient mounting and vacuum further complicates this already complex system. Many calculations can be made to predict some frequencies that may be expected in this system and one could spend considerable time making a mathematical analysis to attempt to identify discrete frequencies actually obtained with the analyzer. Although it is felt that such a mathematical analysis is not justified, one would be remiss if a brief analysis of the frequency spectra were not made.

Air Column Frequency of the Tank

One would expect to find a fundamental air column frequency of the tank. Considering the tank as a closed tube, one might predict the air column vibrations by [6]

$$f = nc/2L \tag{1}$$

where

f = frequency, cps

n = mode of vibration or harmonic

c = velocity of sound in air, fps

L = length of tube, feet

Using this formula one obtains 290 cps for the fundamental frequency. A close examination of the tank shows that the driver actually takes up a considerable volume, Fig. 1, forming several air chambers and now one would expect the 290 cps value to be high. This is substantiated by Figs. 6, 7, and 8.

The fundamental frequency of the vibrating air column can be approximated by reducing the actual air cavity in the tank to an equivalent system and then to a pipe with a length ℓ_k [7], see Fig. 5, where

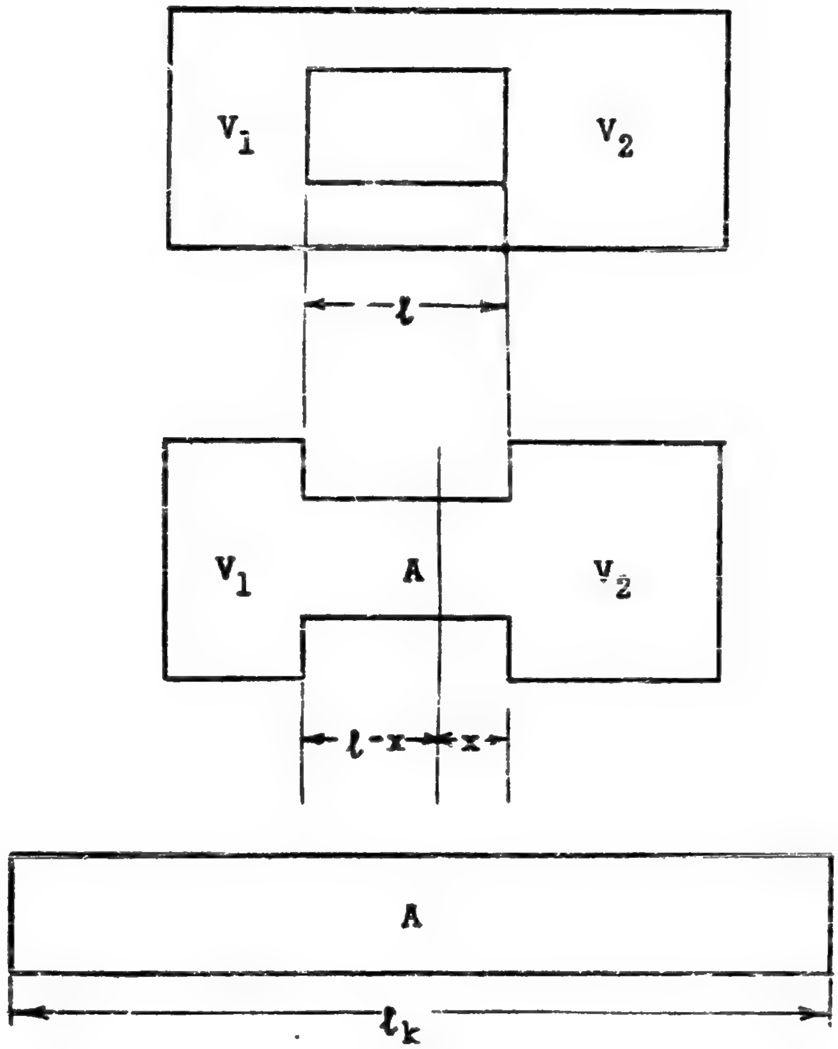
$$\tan\frac{\pi\ell}{\ell_k} = \frac{(A/V_1 + A/V_2)\ell_k/\pi}{1 - \frac{A^2}{V_1 V_2}\frac{\ell_k^{\,2}}{\pi^2}} \tag{2}$$

Solving for ℓ_k

$$\ell_k = 26.25 \text{ inches}$$

Using Raleigh's length correction factor, $\Delta\ell$ [7]

$$\Delta\ell = \frac{\pi r}{4} \tag{3}$$



A = area of connecting pipe

Fig. 5 Schematic Representation of Air Cavity in Tank

$$\Delta\ell = 3.8 \text{ inches}$$

Now solving for f from (1)

$$f = \frac{nc}{2L} = \frac{1126}{\frac{2(26.25 + 3.8)}{12}} = 224.8 \text{ cps}$$

Also, one may approximate the fundamental frequency of the vibrating air column by considering the air cavity equal to two Helmholtz resonators where the frequency of the Helmholtz resonator [8] is: (see Fig. 5)

$$n = \frac{1}{2\pi}\sqrt{\frac{c^2A}{\ell_e V}} \tag{4}$$

where

$$\ell_e = \ell + 0.8\sqrt{A} \tag{5}$$

The frequency of each Helmholtz resonator must be the same. Therefore,

$$n = \frac{1}{2\pi}\sqrt{\frac{c^2A}{[(\ell - x) + 0.8\sqrt{A}]V_1}} = \frac{1}{2\pi}\sqrt{\frac{c^2A}{(x + 0.8\sqrt{A})V_2}} \tag{6}$$

Solving

$$[(\ell - x) + 0.8\sqrt{A}]V_1 = (x + 0.8\sqrt{A})V_2$$

$$x(V_1 + V_2) = \ell V_1 + 0.8\sqrt{A}(V_1 - V_2)$$

$$x = \frac{\ell V_1 + 0.8\sqrt{A}(V_1 - V_2)}{V_1 + V_2} \tag{7}$$

Solving for x and substituting in (6)

$$n = 205 \text{ cps}$$

Thus, by two approximations one obtains 224.8 cps and 205 cps for the fundamental frequency of the air column in the tank.

Looking at Figs. 6 and 7 this frequency appears to be 215 cps and on Fig. 8 as 220 cps. Fig. 6 also indicates a mechanical resonance with a vacuum at 215 cps. One can also identify the 3, 4, 7, 11, 20, 27, and 30 harmonics. Likewise, many air-column harmonics can be identified in Figs. 7 and 8 with air in the tank. Fig. 8 indicates another air-borne frequency at 270 cps. Since each frequency was excited separately, an air-column resonance could have developed at 270 cps.

Steel Diaphragm Vibrations

The steel diaphragm can vibrate with many modes indicated by its nodal circles and nodal diameters. No doubt a nodal diameter will develop if the driver is not symmetrically located. One can approximate some of these frequencies with equation [9, 10]

$$2\pi f = \frac{\alpha}{R^2}\sqrt{\frac{gD}{\gamma h}} \tag{8}$$

and

$$D = \frac{Eh^3}{12(1 - \eta)}$$

where α = constant depending on mode of vibration

g = 386 inches per second squared

γ = weight per unit volume, pounds per cubic inch

h = thickness of plate, inches

E = modulus of elasticity

η = Poisson's ratio

R = radius of diaphragm, inches

The inside diameter of the tank is 11 inches and the clamping ring inner diameter is 13 inches.

The frequency for no nodal diameters or circles is 345 cps using the 13 inch clamping diameter. Fig. 7 indicates a plate frequency at 340 cps when the tank is evacuated. The 560 cps peak on Fig. 6 appears to be for one nodal diameter without a nodal circle. This would suggest a 277 cps frequency for no nodal diameters or circles and one notes a slight peak at this frequency. Another plate frequency can be identified at 1075 cps. Calculations could be continued to identify other plate frequencies.

Frequency in Tank Shell

The fundamental frequency in the vertical portion of

the steel tank can be calculated by using equation (1). This value is approximately 4250 cps. Structure-borne vibration readings are indicated in Figs. 6, 7, and 8 from 4200 cps to 4400 cps. No doubt other frequencies exist in this steel tank shell.

When a vacuum was developed in the tank, it was of the order of less than one millimeter of mercury, absolute. This introduces high stresses in the steel tank, especially in the steel diaphragm. Calculations show that these stresses may be high enough to create a nonlinear system. If this were true it would be difficult to identify discrete frequencies of such a system.

The above analysis gives some idea of the complexity of the vibrating system involved in these tests. The tests were a success in that the importance of air-borne to structure-borne vibrations was determined for the test apparatus used.

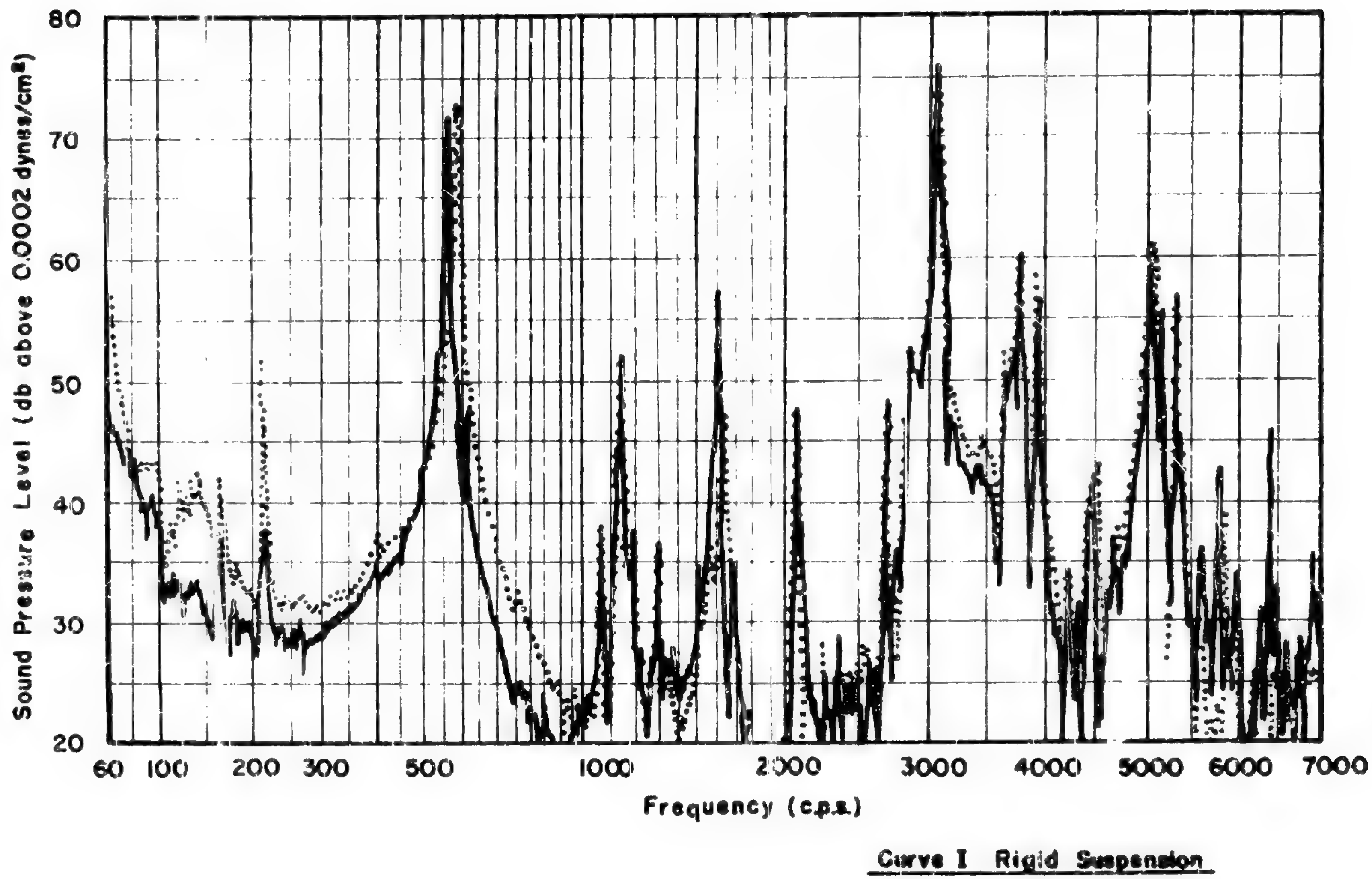
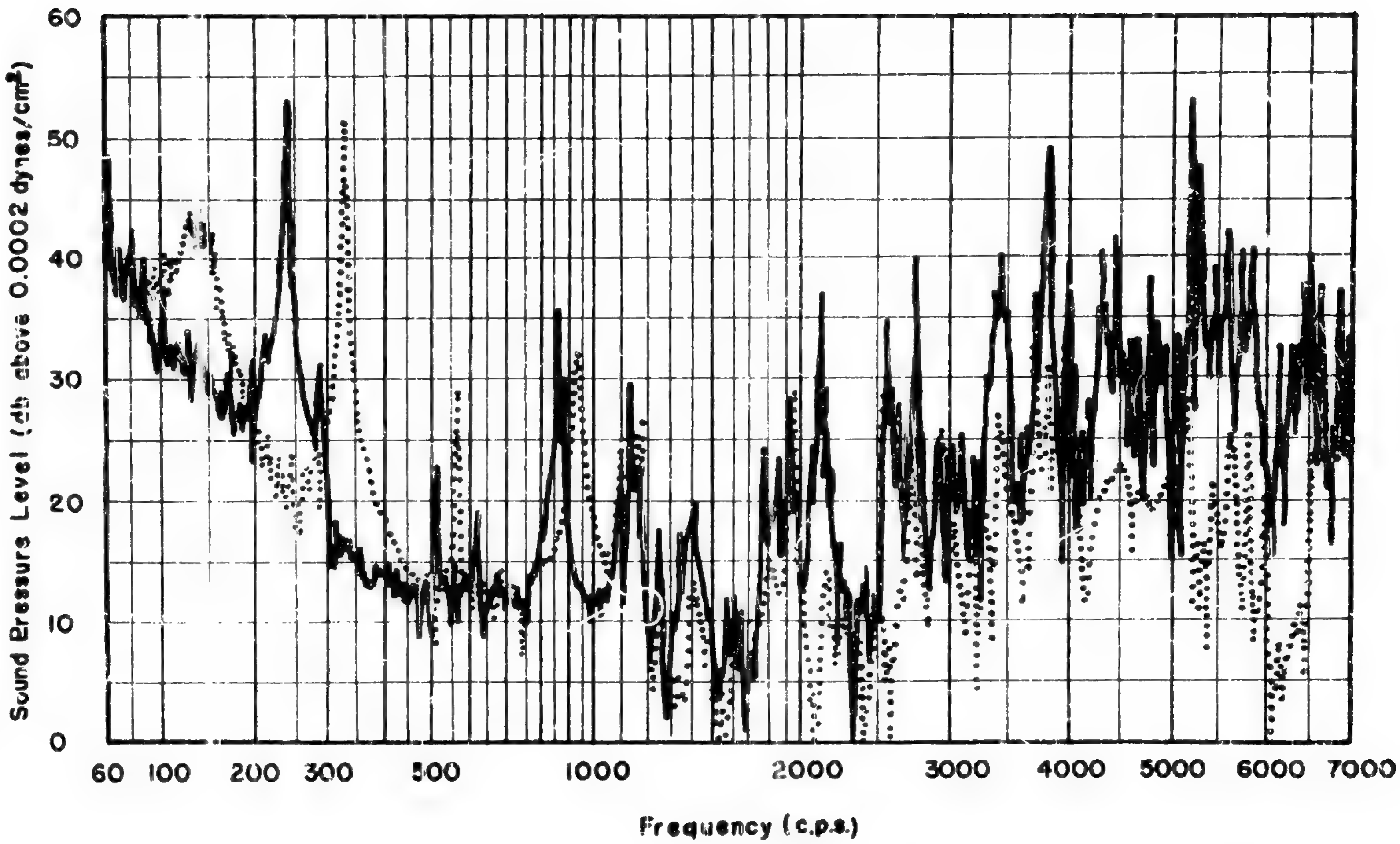
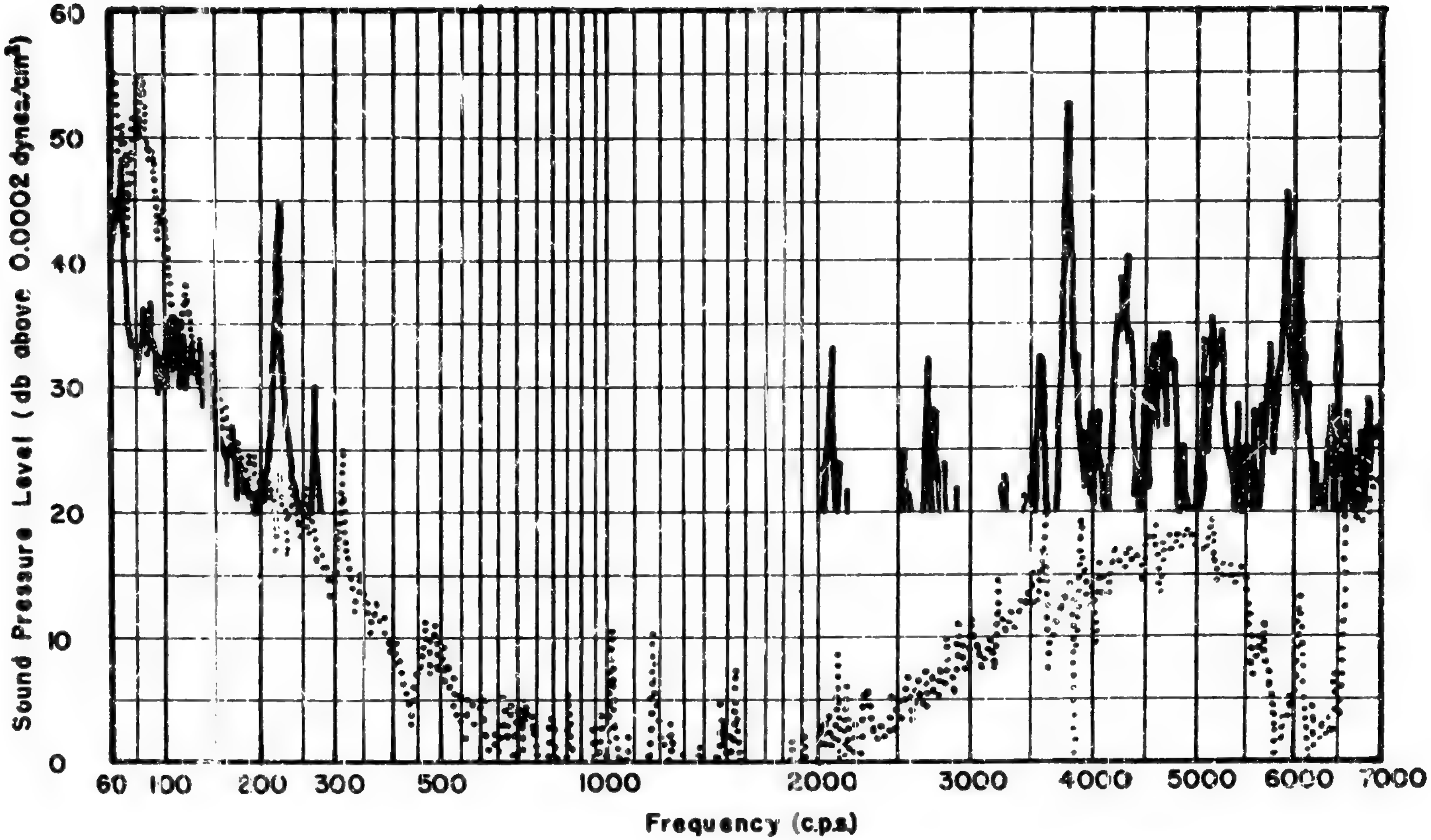

80
70
60
50
40
30
20
Sound Pressure Level (db above 0.0002 dynes/cm²)
60
100
200
300
500
1000
2000
3000
4000
5000
6000
7000
Frequency (c.p.s.)
Curve I Rigid Suspension

Figure 6



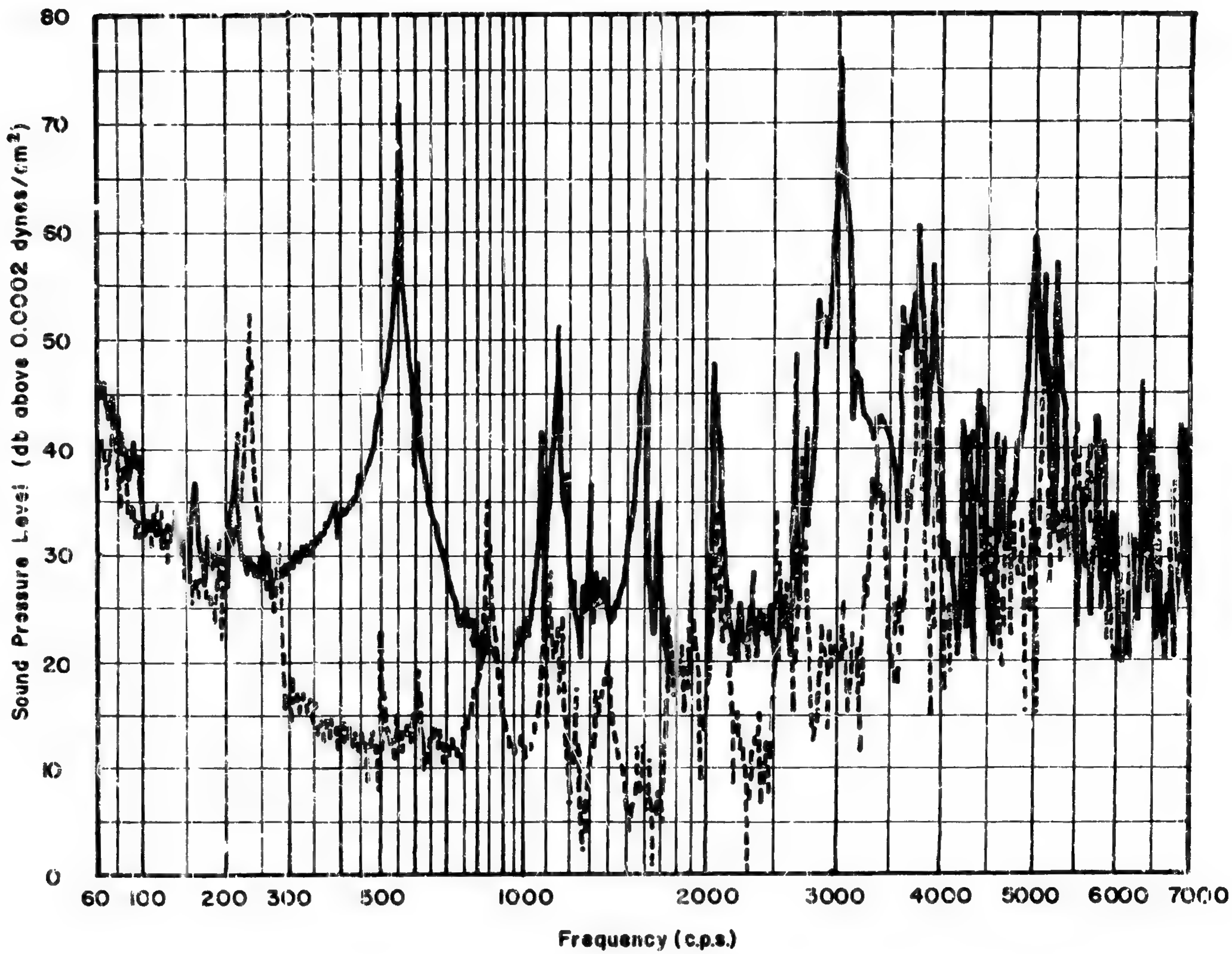
Curve II Resilient Suspension

Figure 7



Curve III Wire Suspension

Figure 8



Curve IV Composite curves I & II

Figure 9

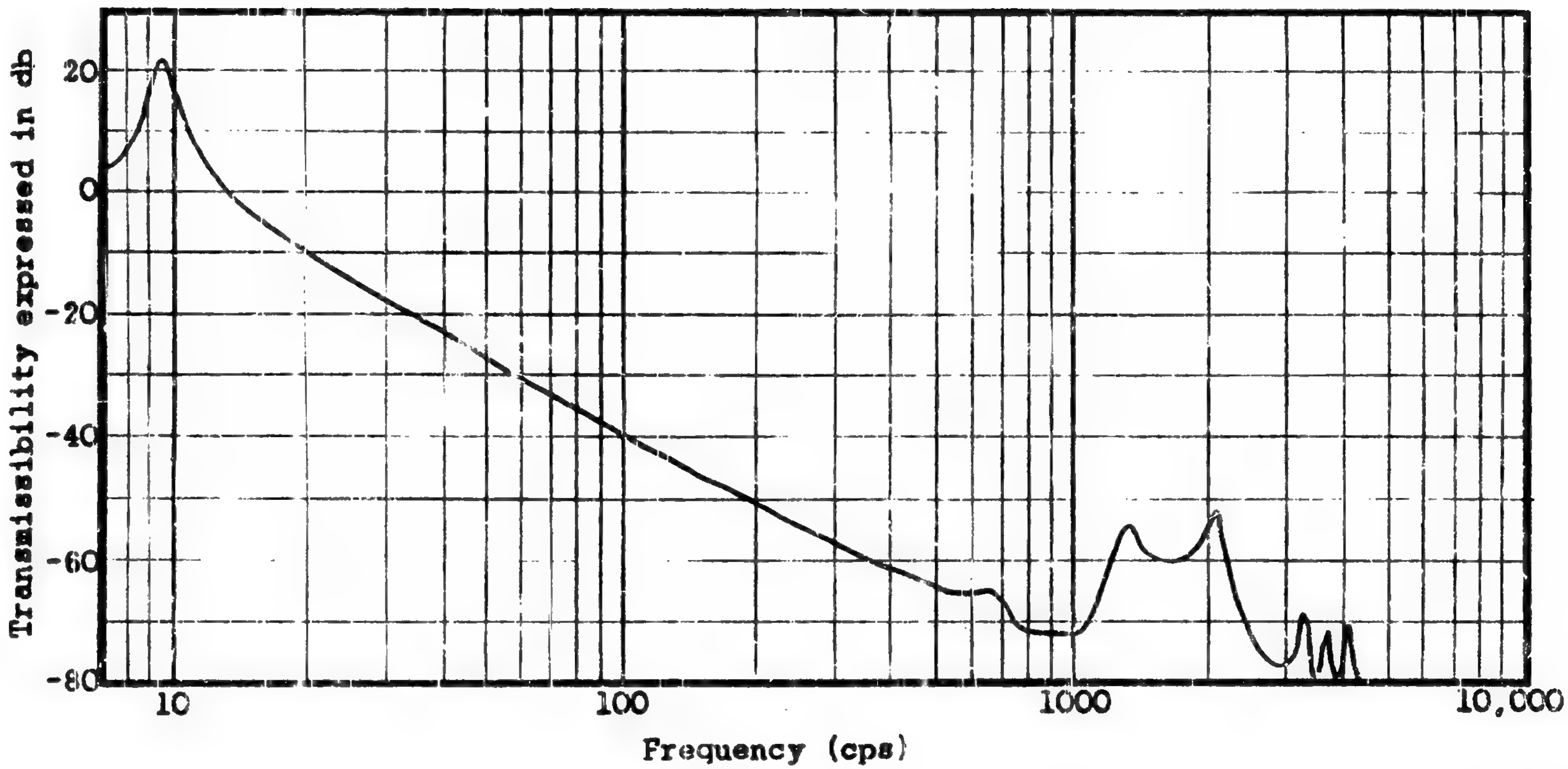


Fig. 10 Transmissibility vs Frequency Curve
for Lord 2C4PH35 Mounting with 42 Pound Load

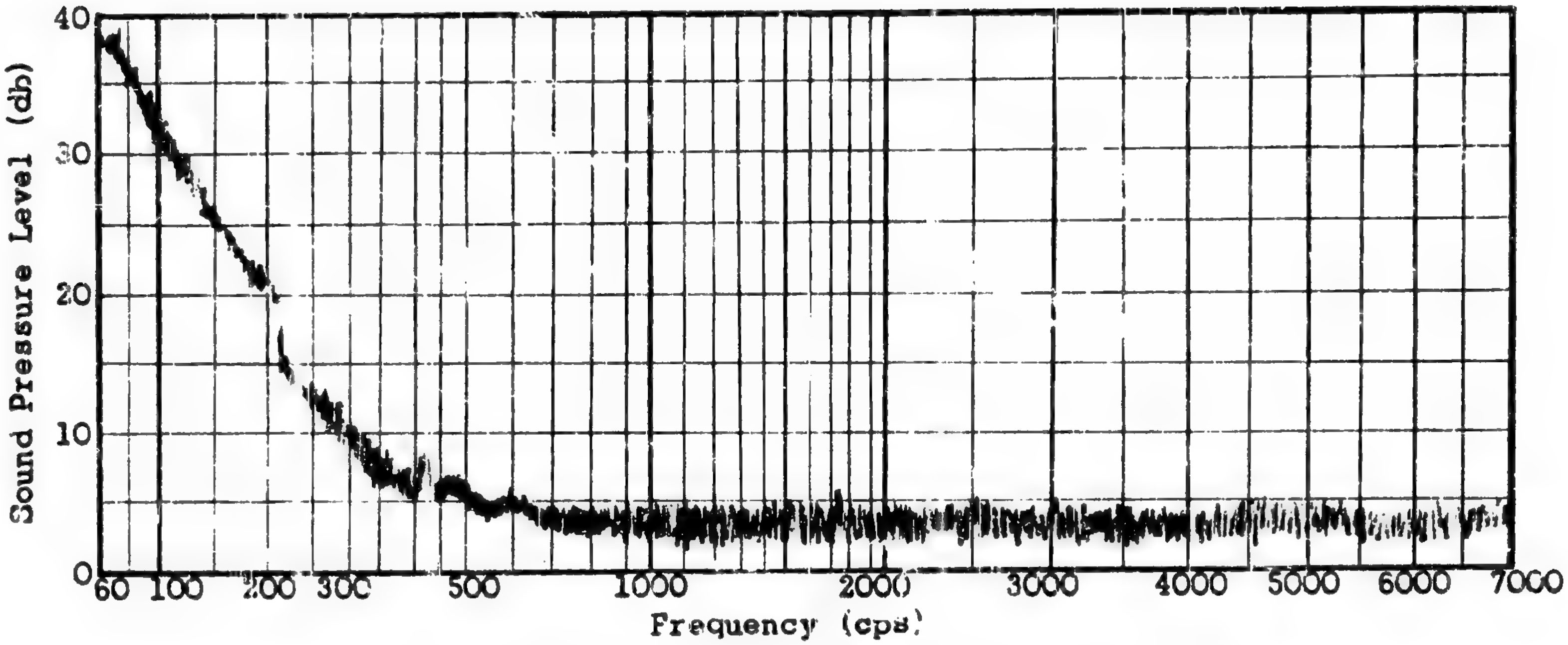


Fig. 11 Frequency vs Sound Room Noise Intensity

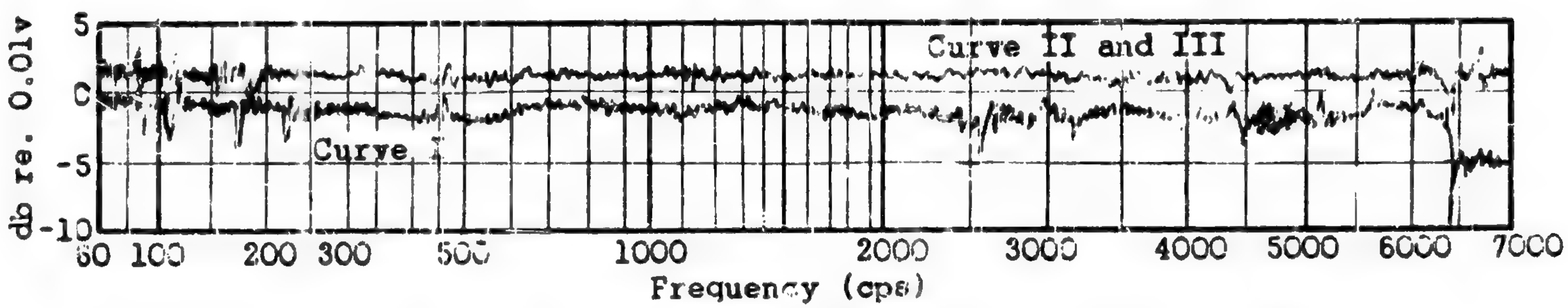


Fig. 12 Frequency vs Acceleration of Exciter Voice Coil when Recording Curves I, II and III

VI. CONCLUSIONS

1. When the vibration exciter or driver was mounted solidly to the steel test tank, the air-borne vibrations were too low in intensity to affect the decibel readings caused by the structure-borne vibrations. The air-borne vibrations had no effect on the readings.

2. The resilient mounting greatly attenuated the structure-borne vibrations.

3. Air-borne vibrations by-passed the resilient mounting at high frequencies. Using the resilient connection, the air-borne vibrations had little effect on the structure-borne vibrations up to 2000 cps, but from 2000 to 7000 cps the air-borne vibrations were predominate, raising the decibel level above those for the structure-borne vibration alone. This is attributed to several factors: (a) air-coupled sound energy by-passing the resilient mounting, and (b) loss of isolation effectiveness of the resilient mounting, causing amplification due to resonance of some part of the tank.

4. Using the resilient mounting or the wire suspension connections, it was found that the resilient mounting did not attenuate structure-borne vibrations above 2000 cps as one would normally expect from basic vibration theory.

5. The ambient noise masks signals from the test tank below 150 cps, and no conclusions can be made in this frequency range.

Although this study is being continued by Illinois Institute of Technology, the findings outlined herein should be correlated with results from other laboratories. This may lead to important contributions in naval design for a quiet operating submarine.

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